UNCLASSIFIED
SECURITY CLASSIFICATION OF THIS PAGE

AD-A202 902

REPORT DOCUMENTATION PAGE

DTIC SELECTE DEC 6 1988 S D

DTIC FILE COPY

(2)

1a. REPORT SECURITY CLASSIFICATION Unclassified	1b. RESTRICTIVE MARKINGS
2a. SECURITY CLASSIFICATION AUTHORITY 2b. DECLASSIFICATION / DOWNGRADING SCHEDULE	3. DISTRIBUTION / AVAILABILITY OF REPORT Approved for public release Distribution unlimited
4. PERFORMING ORGANIZATION REPORT NUMBER(S) D C8	5. MONITORING ORGANIZATION REPORT NUMBER(S) AFOSR-TR- 88-1290
6a. NAME OF PERFORMING ORGANIZATION North Staffordshire Polytechnic — 6b. OFFICE SYMBOL (If applicable)	7a. NAME OF MONITORING ORGANIZATION European office of Aerospace Research and Development/LTS
6c. ADDRESS (City, State, and ZIP Code) Dept of Mechanical & Computer Aided Engineering Beaconside Stafford ST18 0AD United Kingdom	7b. ADDRESS (City, State, and ZIP Code) [redacted] Same as 8c.
8a. NAME OF FUNDING / SPONSORING ORGANIZATION Air Force Office of Scientific research — 8b. OFFICE SYMBOL (If applicable) NA	9. PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER AFOSR 84-0368
8c. ADDRESS (City, State, and ZIP Code) AFOSR/NA Bldg 410 Bolling AFB / DC 20332 - 6448	10. SOURCE OF FUNDING NUMBERS PROGRAM ELEMENT NO. 61102F — PROJECT NO. 2302 — TASK NO. B1 — WORK UNIT ACCESSION NO.

11. TITLE (Include Security Classification)
Vibration Control in Rotating Machinery using Variable Dynamic Stiffness Squeeze-Films

12. PERSONAL AUTHOR(S)
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13a. TYPE OF REPORT	13b. TIME COVERED	14. DATE OF REPORT (Year, Month, Day)	15. PAGE COUNT
Final Scientific	FROM Sept.84 TO Mar.88	June 1988	

16. SUPPLEMENTARY NOTATION

17. COSATI CODES			18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number)
FIELD	GROUP	SUB-GROUP	Bearing, Squeeze Film, Vibration, Rotors

19. ABSTRACT (Continue on reverse if necessary and identify by block number)

This report describes work carried out with the aim of developing a combined hydrostatic and squeeze-film bearing, for rotating machinery, whose dynamic characteristics may be tuned during operation of the machine. The purpose of this is to enable the operator to exercise control over machine critical speeds and vibrations.

A computer program has been written to predict the characteristics of the bearing type, the program allows for the presence of accumulators linked to the bearing oil film whose purpose is to modify the bearing dynamic characteristics. A test rig has been designed and built, based on a General Electric TF34 turbofan engine, and both theoretical and experimental results confirm that a substantial shift in critical speed is effected by using the bearing, and that system vibration and force transmissibility may be reduced substantially when compared with the performance of conventional squeeze film bearings and journal bearings.

Keywords; vibration, Rotors

20. DISTRIBUTION / AVAILABILITY OF ABSTRACT ☐ UNCLASSIFIED/UNLIMITED ☒ SAME AS RPT ☒ DTIC USERS	21. ABSTRACT SECURITY CLASSIFICATION Unclassified
22a. NAME OF RESPONSIBLE INDIVIDUAL Dr. Anthony Amos	22b. TELEPHONE (Include Area Code) (202) 767-[redacted] 4937 — 22c. OFFICE SYMBOL AFOSR/NA

DD FORM 1473, 84 MAR — 83 APR edition may be used until exhausted. All other editions are obsolete.

SECURITY CLASSIFICATION OF THIS PAGE
UNCLASSIFIED

AFOSR-TR- 88-1290

Final Technical Report on Project Titled

VIBRATION CONTROL IN ROTATING MACHINERY USING VARIABLE DYNAMIC STIFFNESS SQUEEZE FILMS.

On Work Carried Out Under USAF Grant AFOSR 84-0368

APRIL 1985 TO JULY 1988

Written By: Dr M.J.GOODWIN.
M.P.ROACH.

1.0 INTRODUCTION

The following report presents the areas of investigation and the conclusions of the research program on the AFOSR grant 84-0368 for the complete grant period April 1985 to June 1988. The specific areas of investigation were :-

1.0 Theoretical Study of Bearing Characteristics Including:-

- Theoretical analysis of hydrostatic support static and dynamic characteristics.
- Theoretical analysis of squeeze-film effects of hydrostatic land area.
- The effect of the connection of nitrogen bag accumulators to the hydrostatic bearing recess, via flow restrictors, on the support characteristics.

2.0 Theoretical Study of Effect of Proposed Bearing Design on Rotor-Bearing System Dynamics Including:-

- Theoretical study of the effect of support impedance on the steady-state response to unbalance of a symmetric multi mass rotor.
- A study of the effect of support impedance on the force transmitted through the supports of rotating machines.

3.0 An Investigation into the Effect of the Proposed Bearing Design on System Stability Including:-

- the effect of variation of support impedance and rigid rotor critical speed.

- recommendations for the best overall bearing characteristics

4.0 Experimental investigation into the above areas using an experimental rig designed as a model of the G.E. TF34 turbofan engine running on rolling element bearings each supported by hydrostatic squeeze film systems.

A summary of each of the above areas investigated is provided, together with a brief summary of the results obtained from the investigation. A list of references is also included, containing technical publications which document these areas of research.

1.0 Theoretical Study of Bearing Characteristics

By writing expressions for lubricant flow into and out of a hydrostatic bearing recess and equating these flows, expressions were derived which describe the load carrying capacity, stiffness and lubricant flow rate for the hydrostatic bearing. These expressions were a function of the steady eccentricity of the shaft centre within the bearing clearance. This analysis was then extended, by using a pertubation method, to yield values of hydrostatic bearing stiffness and damping with the effects of both lubricant compressibilty and lubricant inertia accounted for. Using further extensions of the recess flow analysis, the affect of the connection of accumulators to the recess via flow restrictors was investigated. Further details of these analyses are given in references (1-4). The bearing design investigated consisted of four hydrostatic pads equispaced around the bearing circumference and with 360° circumferential lands for the purposes of providing adequate squeeze-film damping.

A computer program based on the above theory was written and used to investigate the following:

- Variation of bearing eccentricity with applied static load.
- Variation of static stiffness with load eccentricity.
- Variation of lubricant flow rate with eccentricity.
- Variation of bearing dynamic stiffness and damping with eccentricity.
- Variation of the squeeze-film damping generated at the bearing lands with eccentricity.

- Effect on the overall bearing stiffness and damping of connecting accumulators to the bearing recess via variable flow restrictors.

- Effect of supply pressure on steady load carrying capacity, static stiffness and dynamic stiffness and damping.

- Effect of the ratio recess pressure/supply pressure on steady load carrying capacity, static stiffness and on the dynamic stiffness and damping.

- Effect of lubricant inertia and compressibility on bearing dynamic stiffness and damping.

The following conclusions were reached as a consequence of the above investigations:-

- The values of hydrostatic bearing static stiffness which can be produced are sufficiently high to result in a negligible steady eccentricity of the shaft centre within the bearing clearance. This was achieved with a supply pressure of 2.0 MPa and bearing dimensions of 35mm pad width x 120mm diameter carrying loads of 250N. This resulted in an effective static stiffness of 35 MN/m.

- The stiffness of the bearing under the action of a dynamic load is the same as that when only a static load is applied (although this is only the case in the absence of accumulators or when the lubricant inertia effects are not excessive).

- The hydrostatic bearing impedance may be effectively represented by linear coefficients when the shaft vibration amplitude at the bearings is less than half film radial clearance.

- Full 360° circumferential lands can be used to provide sufficient squeeze-film damping to the system at the bearing locations to attenuate the steady-state response of the system and improve the systems stability (see sections 2.0 and 3.0).

- The squeeze film damping coefficient is essentially linear for shaft vibration amplitudes less than half the radial film clearance but rapidly becomes very non-linear for larger amplitudes.

- If the shaft is centralised wihtin the bearing film clearance the hydrostatic and squeeze-film cross-coupling coefficients are zero. For small values of shaft eccentricity the cross-coupling coefficients can assume non zero values but are still very amall compared to the direct terms and so they do not have a significant effect on the overall bearing characteristics.

- Significant values of bearing damping are produced by hydrostatic action at the bearing (provided that cavitation does not occur) in the absence of accumulators being connected. This damping complements that which is produced by squeeze-film action.

- The optimum value of the ratio recess pressure/supply pressure to maximise the bearing static stiffness is 0.5.

- Due to the generally large values of bulk modulus of commercial oils, the effect of lubricant compressibility on bearing characteristics is negligible.

- With the use of capillary compensation at the supply line, oil viscosity (and therefore temperature) variations do not affect the bearing load carrying capacity. The bearing damping produced however, varies in proportion to the oil viscosity.

- The bearing damping always increases with increasing frequency and with increasing supply and accumulator line fluid inertia.

- Connecting accumulators to the bearing recesses does not affect the bearing static load carrying capacity and therefore the hydrostatic action can still be used to centralise the shaft within the bearing clearance.

- The dynamic stiffness and damping of the bearing both reduce as a result of accumulator attachment by an amount which is dependent upon the value of the flow resistance between the accumulator and the recess.

- With infinite accumulator line resistance the accumulator has no effect on the stiffness of the pad but with negligible flow resistance the supports dynamic stiffness and damping can be reduced to near zero whilst still maintaining shaft centralisation within the bearing film clearance. Thus by installing remote control valves between the accumulators and recess it is possible to change the support dynamic stiffness during system runup to operating speed if required.

- Inertia of the lubricant in the supply line restictor and in the accumulator line restrictor can result in either dynamic stiffness increase or decrease with increasing frequency depending upon the magnitude of inertia coefficient. For realistic inertia values however these effects are negligible.

2.0 Theoretical Study of the effect of the Proposed Bearing Design on Rotor-Bearing System Dynamics.

The theory used to describe the rotors steady state vibration response is a transfer matrix method similar to that described by Rao (Wiley 1983)(5). For the present investigation a symmetric five mass lumped parameter model was used to represent the rotor with the majority of the mass concentrated at the shaft midspan and with significant masses associated with each bearing station. The remaining two masses model the shaft mass between these midspan and bearing station points. The model is also described in reference (2).

A computer program based on the above theory was used to investigate the response of the model to unbalance. The input data for the program relating to the bearing characteristics was obtained from the investigation described in section 1.0 of this report. The response program was used to investigate the following:

- The bearing and shaft midspan response to unbalance at the shaft midspan.

- The bearing and shaft midspan response to unbalance at the bearing stations.

- The force transmision through the bearings with shaft midspan unbalance.

- The force transmission through the bearings with bearing station unbalance.

- The variation of the machine critical speeds with bearing stiffness decrease from the rigid bearing case.

- The effect of the magnitude of bearing damping introduced into a flexible bearing system on the shaft midspan and bearing response.

- The effect of the magnitude of bearing damping introduced into a flexible bearing system on the force transmission through the bearings.

The specific values of bearing stiffness used in the above investigations were chosen to represent the effect on rotor behaviour of (1) a very high stiffness support system, (2) an orthodox centralised squeeze film support system, (3) a tuned support system where the supports are used as dynamic vibration absorbers and (4) the hydrostatic squeeze film system with direct accumulator attachment ie. no accumulator flow resistance. The ratios K of the bearing stiffness to shaft stiffness were therefore as follows:-

1) Very high stiffness bearing	$K = 1 \times 10^8$
2) Orthodox centralised squeeze film, typically	$K = 0.5$
3) 'Tuned' supports K = M	$K = 0.15$
4) Hydrostatic squeeze film ('full' accumulator)	$K = 0$

where M is the ratio effective bearing mass/effective rotor midspan mass.

As a consequence of carrying out the above investigations the following conclusions where reached:

- With rigid supports the system behaves essentially as a single degree of freedom system within the speed range reached (0 to 4xrigid support critical speed).

- The introduction of flexible supports results in the single critical speed peak of a rigidly supported rotor being replaced by several critical speeds. These critical speeds include two 'rigid' body modes, the lower one being a translational mode and the higher one being a conical mode. The conical mode does not form if the unbalance present is symmetric about the shaft midspan.

- A third resonance peak, which is the shaft first flexible mode, may be introduced into the operating speed range if the bearing flexibility is reduced to a low enough value (for example, if the operating speed is in excess of twice the rigid bearing critical speed in the case of systems which have a mass ratio of M = 0.15).

- The introduction of bearing damping can attenuate the rotor and bearing responses at all of the critical speeds, provided that the magnitude of the damping is not excessive. If the bearings are overdamped the system tends to behave more like a system with rigid bearings with a single resonace peak at the rigid bearing critical speed.

- For a given system and bearing stiffness, an optimum value of bearing damping exsists which minimises the rotor vibration amplitudes over the speed range of interest and also minimises the force transmitted by the bearings at the rotor critical speeds.

- When the optimum support damping value is present the maximum rotor vibration amplitude ratio over the speed range of interest is determined by the stiffness ratio K and mass ratio M. The tuned support system (K=M) yields the lowest rotor midspan amplitudes over the speed range and higher bearing stiffnesses cause a significant increase in response amplitudes. If however the bearing stiffness is reduced below this tuned value, which results in transmissibility reductions and stability improvements (see section 3.0), there is a negligible increase in the rotor vibration provided that the system mass ratio is low.

- With light bearing damping, reducing the support stiffness to zero removes the rigid body modes from the response plot and consequently the first critical speed encountered is the shafts first free-free critical speed. The ratio of the free-free critical speed to the rigid bearing critical speed is determined only by the system mass ratio but is always greater than one. Bearing damping attenuates the associated response and provided that the bearing damping is not excessive, also reduces the speed at which it occurs.

- For flexible bearing systems, with shaft midspan unbalance, the bearing response at the rigid bearing critical speed is always equal to the unbalance mass eccentricity of the system. This infers that the radial film clearance of the bearing must always be greater than this unbalance eccentricity and infact greater than twice this value if the system is to operate within the squeeze film linear damping range.

- High support stiffness values increase the supercritical force transmissibility and lowers the effectiveness of damping in attenuating the rotor vibration amplitude. For this reason, bearing stiffness increases should not be used as a means of moving critical speeds away from the machine operating speed.

- At post critical speeds the rotor and bearing response tends towards the unbalance eccentricity in the respective plane.

- With zero bearing stiffness the transmitted force can be kept below the force level generated by the unbalance, provided that optimum damping is present.

- All critical speeds may be avoided by bearing stiffness changes during system runup provided that sufficient change in bearing stiffness are used.

- Regarding the systems response to unbalance, it is always beneficial to provide optimum bearing damping. The best bearing stiffness ratio to adopt is that where the bearing stiffness ratio is equal to or less than the system mass ratio. It is however advantageous to reduce the bearing stiffness to zero as the 'rigid body' critical speeds are removed from the reponse plot and the response amplitude peaks and force transmissibility peaks associated with these critical speeds are then absent.

- Bearing stiffness switching during system runup is advantageous with regards to avoiding the resonant peak which is associated with shaft critical speeds if the bearing damping available is very low.

- The removal of the bearing dynamic stiffness in the presence of optimum support damping results in a 60% reduction in peak rotor vibration amplitude and a 65% reduction in force transmissibilty compared to that resulting from the use of conventional centralised squeeze-film bearing (4). The bearing force transmissibility for zero stiffness bearings with optimum damping is always less than one.

Futher details of the steady-state response of the system investigated and of the analysis techniques used can be found in references (4) and (6).

3.0 Investigation into the Effect of the Proposed Bearing Design on System stability.

The steady state response analysis discussed in section 2.0 of this report, assumes that the system is stable. In the absence of stability the rotor, when displaced from its equilibrium position, has a vibration which does not decay to the equilibrium orbit but increases in amplitude to an unbounded orbit resulting in violent machine vibrations and possibly machine failure. Several causes of instability have previously been identified, namley rotor internal friction, bearing or support cross-coupling stiffness and aerodynamic cross-coupling.

A computer program based on the Routh Hurwitz stability criteria was used to examine the stability of a symmetrical single rotor mass system which had a flexible shaft and ran in flexible damped bearings. The equations of motion for such a system can be developed (7,8) to yield a system characteristic equation, the roots of this equation usually occur in complex conjugate pairs with magnitude of the largest real root indicating the degree of stability (or instability) present. The variation of this root with various system parameters was therefore used to indicate beneficial design trends which promote the stability of rotating machines. The program was used to investigate the following:

- Variation of stability with bearing damping.

- Variation of stability with bearing stiffness in the presence of optimum bearing damping.

- Variation of stability with aerodymamic cross-coupling coefficient.

As a consequence of the above investigations the following conclusions were reached:

- In the absence of aerodynamic cross-coupling, bearing cross-coupling and rotor internal friction, the system is always stable.

- For a given bearing impedance, the degree of stability of a rotating machine is proportional to the shaft stiffness and is inversly proportional to the effective rotor mass. For this reason the use of highly flexible shafts in rotating systems has a tendancy to promote instability.

- For any given bearing stiffness there is an optimum value of bearing damping which maximises the systems stability. The bearing damping required to maximise a systems stability increases with increasing bearing stiffness.

- Introducing damped flexible bearings into a rotating system increases the stability of the system, with the stability increasing as the support stiffness is decreased provided that optimum support damping is present. At a bearing to shaft stiffness ratio of $K = 0.1$ the stability of the system with optimum bearing damping is maximised. Further bearing stiffness reductions below this value result in reduced system stability.

- Aerodynamic cross coupling always has a destabilising effect and therefore causes stability reductions regardless of the bearing impedance present. There is a limit to the value of aerodynamic cross coupling that may be offset to ensure stability by the modification of the bearing impedance. When this amount of aerodynamic cross-coupling is present or has been exceeded the only means of stabilising the system are to reduce the running speed (the magnitude of aerodynamic cross-coupling being dependent upon the running speed) or replace the shaft with one of higher stiffness. The bearing/shaft stiffness ratio of $K = 0.1$ which maximises the system stability with optimum damping still applies however, regardless of the shaft stiffness value present, although the optimum damping value is dependent upon the absolute value of the shaft stiffness as well as the bearing/shaft stiffness ratio.

- For a typical machine the most appropriate bearing characteristics to maximise the machines stability are a bearing/shaft stiffness ratio of $K = 0.1$ with optimum support damping for stability.

- The optimum damping for stability is very close to the optimum damping for steady-state response to unbalance. Increases in rotor response due to the use of optimum damping for stability, or alternatively reductions in stability due to the use of optimum damping for steady-state response, are both negligible.

From the above steady-state, transmissibility and stability analyses the overall recommendations on support design are:

- For steady-state rotor/bearing response $K = M$
  $C = C_{optimum\ steady\text{-}state}$

- For bearing force transmissibility $K = 0.0$
  $C = 0.0$

- For stability $K = 0.1$
  $C = C_{optimum\ stability}$

where C is the bearing damping coefficient.

From the conclusions presented in sections 2.0 and 3.0 it follows that the bearing characteristics which optimise the rotor vibration, bearing force transmissibility and are still consistent with adequate system stability are therfore:

- $K = 0$
  $C = C_{optimum\ steady\text{-}state}$

with support stiffness switching to $K = 0.1$ at operating speed if instability problems arise. These bearing characteristics may be achieved using the hydrostatic squeeze film system with accumulators installed.

4.0 Experimental results

Experimemtal results were obtained from the test rig for the forced unbalance response of the shaft at the rotor and bearing positions and the free vibration (transient) response at the rotor and bearing positions. The two bearing/shaft stiffness ratios investigated (and also the damping values used) were:

- $K = 3.0$ (no accumulators)
  $C = 1.1\ C_{optimum}$

- $K = 0.002$ ('full' accumulator.)
  $C = 1.4\ C_{optimum}$

The experimental results confirm the above theoretical conclusions within accepted limits of experimental error, both for the steady-state response and also for stability.

The stiffness ratios $K = 0.002$ and $K = 3.0$ resulted in magnification factors of 2.0 and 16 respectively at the rotor; the lower of these obtained with accumulators connected to the bearing, represents a vibration level which is 60% less than that obtained when conventional squeeze-film bearings are used. Further reductions in vibration amplitude would result if the bearing damping present for $K = 0.002$ had been equal to the optiumum damping value.

An indication of the system stability was obtained by subjecting the rotor to a force impulse and observing the time required for the resulting transient to decay (and observing the number of cycles of vibration). It was found that the transient decay times were reduced by a factor of 4.2 with a bearing stiffness change from $K = 3.0$ to $K = 0.002$.

The above experimental results were obtained at rotational speeds upto 6000rpm; the theoretical results show that the system first critical speed had been exceeded (confirmed by experimental vibration phase readings) and that above this speed the system response steadily reduces, tending to zero at the bearings and the systems unbalance eccentricity at the rotor. Further details of the experimental investigations may be found in references (9) and (10)

List of Technical Publications

1. M J Goodwin, M P Roach — 'Vibration Control in Rotating Machinery Using Variable Dynamic Stiffness Squeeze-Films'
   1st Full Interim Scientific Report - March 1986, Vol 2.
   AFOSR Grant 84-0368.

2. M J Goodwin, M P Roach — 'Variable Dynamic Squeeze-Films for Rotating Machinery'
   2nd Interim Report - October 1986
   AFOSR Grant 84-0368.

3. M P Roach — 'Variable Stiffness Squeeze-Film Supports for Rotating Systems'
   MPhil/PhD Transfer Report - January 1987

4. M J Goodwin, M P Roach, J Penny — 'The Elimination of Shaft Critical Speeds Using Parallel Hydrostatic and Squeeze-Film Bearings'
   Presented at ASME Sponsored Symposium on Transport Phenomena. Dynamics and Design of Rotating Machinery
   Hawaii - April 1988.

5. J S Rao — 'Rotor Dynamics'
   Wiley Eastern 1983.

6. M J Goodwin
M P Roach
'The Analysis of Combined Squeeze-Film and Variable Stiffness Hydrostatic Bearings, and Their Use in Aircraft Engine Vibration Control'
To be presented at I.Mech.E Sponsored Conf. Vibrations in Rotating Machinery.
Edinburgh - September 1988.

7. M J Goodwin
P J Ogrodnik
M P Roach
'Optimised Support Impedance for the Stabilization of Rotating Machinery'
(In Preparation)

8. M J Goodwin
P J Ogrodnik
M P Roach.
'The Effect of Bearing and Aerodynamic Cross-Coupling on the Stability of Rotating Machinery'
(In Preparation)

9. M J Goodwin
M P Roach
'An Experimental Investigation of the Effectiveness of Combined Hydrostatic Squeeze-Film Bearings for Controlling Rotor-Bearing System Dynamics'
(In Preparation).

10. M P Roach
'Vibration Control in Rotating Machinery Using Variable Dynamic Stiffness Hydrostatic Squeeze-Films'
PhD Thesis, CNAA (Staffordshire Polytechnic).
(Expected Publication November 1988).